SOME LIMITATIONS IN APPLYING
CLASSICAL EHD FILM-THICKNESS
FORMULAE TO A HIGH-SPEED BEARING

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ABSTRACT

Elastohydrodynamic film thickness was measured for a 20-mm ball bearing using the capacitance technique. The bearing was thrust loaded to 90, 448, and 778 N (20, 100, and 175 lb). The corresponding maximum stresses on the inner race were 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000 psi). Test speeds ranges from 400 to 14 000 rpm. Film thickness measurements were taken with four different lubricants: (a) synthetic paraffinic, (b) synthetic paraffinic with additives, (c) neopentylpolyol (tetra) ester meeting MIL-L-23699A specifications, and (d) synthetic cycloaliphatic hydrocarbon traction fluid. The test bearing was mist lubricated. Test temperatures were 300, 338, and 393 K.

The measured results were compared to theoretical predictions using the formulae of Grubin, Archard and Cowking, Dowson and Higginson, and Hamrock and Dowson.

There was good agreement with theory at low dimensionless speed, but the film was much smaller than theory predicts at higher speeds. This was due to kinematic starvation and inlet shear heating effects. Comparisons

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with Chiu's theory on starvation and Cheng's theory on inlet shear heating were made.

SYMBOLS

a major semi-axis of Hertz contact, m (in.)

b minor semi-axis of Hertz contact, m (in.)

E reduced modulus of elasticity, $\dfrac{1}{E} = \dfrac{1}{2}\left(\dfrac{1 - \nu_1^2}{E_1} + \dfrac{1 - \nu_2^2}{E_2}\right)$

E_1, E_2 Young's modulus for body 1 and 2, GPa (psi)

F normal applied load, N (lb)

G materials parameter, $\alpha E'$

H film thickness, h/R_x

h film thickness, m (in.)

k ellipticity ratio, a/b

R_x equivalent radius in rolling direction, m (in.), $\dfrac{1}{R_x} = \dfrac{1}{R_{x1}} + \dfrac{1}{R_{x2}}$

U speed parameter, $u\eta_0/E'R_x$

u average surface speed, m/sec (in./sec)

W load parameter, $F/E'R_x^2$

x coordinate in rolling direction, m (in.)

y coordinate in transverse direction, m (in.)

1 body 1

2 body 2

α pressure-viscosity exponent, GPa^{-1} (psi^{-1})

η_0 atmospheric viscosity, $N\text{-sec}/m^2$ $(lb\text{-sec}/in.^2)$

ν_1, ν_2 Poisson's ratio for body 1 and body 2

INTRODUCTION

The elastohydrodynamic (EHD) lubrication of bearings has been investigated theoretically and experimentally for more than 30 years [1 to 16]. An EHD film is formed when non-conforming lubricated machine elements are pressed together in rolling and/or sliding contact. In essence, EHD film formation depends on the coupled effects of physical changes in the lubricant, which are caused by high pressures in the contact zone, and elastic changes in the shape of the contact zone which affect the pressure distribution. The high pressures in the EHD contact act to squeeze out the lubricant. However, the lubricant becomes thicker (more viscous) with increasing pressure and resists being squeezed out. The net result is the formation of a thin lubricant film that is beneficial in preventing seizure and rapid wear of contacting parts in motion.

Experimental measurement of EHD film thickness is very difficult because the film is generally less than a micrometer thick. Various methods have been developed and used to measure the EHD film thickness. These are the X-ray, optical and electrical capacitance and conductance methods [17]. X-ray measurements only are suited to measure film thickness between contacting disc rollers. This is because there must be a clear line of sight for the X-rays to pass through the EHD contact zone. Optical methods are generally applied to the case of a steel ball on a transparent disc. However, a recent application was made to a radial loaded roller bearing with a quartz window in the outer race [18]. In [19] it was concluded that film thickness measurement using the

conductance method is not practical. The capacitance technique was used to obtain the film thickness data presented in [20].

Operating EHD parameters of rolling-element contacts are conveniently arranged into dimensionless groupings. These are the load parameter W, the speed parameter U, the materials parameter G, and the film thickness, H. The speed parameter has a stronger effect on film thickness than the load parameter. Most experimental data which have been genrated have been for speed parameters U less than 10^{-10}.

Critical engineering design of mechanical components such as rollent-element bearings and gears requires that engineering data and theory on elastohydrodynamic (EHD) film formation be accurate and comprehensive. The research reported herein, which is based on the work reported in [20], was undertaken to experimentally investigate the elastohydrodynamic film thickness at speed parameter U values greater than 10^{-10}. The primary objectives were to (a) experimentally measure the elastohydrodynamic film (EHD) thickness with four lubricants to speed parameter U values to 10^{-8}, (b) compare the experimental EHD results with those obtained by other investigators and (c) compare EHD measured film thickness measurements with classical film thickness formulae to the practical situation of a 20-mm bore angular-contact ball bearing operating at high speed.

EHD TEST RIG

The experiments were conducted using the EHD film thickness test rig shown in figure 1 and initially described in [20]. The test rig consists of a rotating driveshaft supported by two bearings running

against a test bearing which is located in a vibration isolated housing. The shaft is driven with an air turbine. Shaft speed is sensed with a magnetic pickup. The speed is held constant during a test with an automatic controller. The load is applied pneumatically. The linear ball bushing allows the thrust load to be transmitted from the pneumatic loader through the shaft to the test bearing. The thrust load is determined from the air pressure acting on the loading cylinder.

The temperature of the test bearing is controlled by cartridge heaters which are located in the test bearing support block. The temperatures were held constant within approximately 3 K (5° F) during a test using an automatic controller on the heaters. Temperatures were sensed by thermocouples placed on the bearing outer race and in the inlet lubrication stream. The bearings were mist lubricated with separate systems for support bearings and test bearing. A separate heater allowed the inlet mist to the test bearing to be held equal to the bearing temperature.

Test Bearings

The test bearings for this experiment were 20-millimeter bore angular-contact bearings having three 7.15-millimeter (9/32-in.-) diameter balls. The bearing specifications are listed in table I. The bearing had a polyimide cage and the outer race was electrically isolated from the housing. The spring-loaded thermocouple at the outer race and the slip rings on the drive shaft provided electrical contact for measurements across the test bearing.

Test Lubricants

Four different lubricants were used in the EHD film thickness experiments. The lubricant properties are given in table II. Lubricants A and B were experimental synthetic paraffinics with similar properties, but B had additives whereas A did not. Lubricant C was a 5-centistoke neopentylpolyol (tetra) ester. This is a type II lubricant, qualified to MIL-L-23699 as well as the lubricant specifications of most major aircraft-engine producers. Lubricant D is a synthetic cyclo-aliphatic hydrocarbon having a high coeffient of traction [21]. Lubricants C and D have similar temperature-viscosity properties. However, lubricant D has a larger pressure-viscosity exponent.

The elastohydrodynamic film thickness affects rolling-element fatigue life [22]. In general, the larger the film, the longer the life. Under given operating conditions, the film formed is a function of the initial ambient viscosity of the lubricant and the lubricant's pressure-viscosity coefficient. Comparative rolling-element fatigue tests were conducted with lubricants C and D [23,24]. No statistical life differences between these two fluids were exhibited. A synthetic paraffinic oil was also tested. However, this fluid had a lower viscosity than either lubricants A and B. The lower-viscosity synthetic paraffinic oil also exhibited no statistical life difference when compared to the tetraester fluid (lubricant C) [24]. A higher-viscosity paraffinic lubricant, such as the two of the current study, would be expected to have a larger EHD film thickness and, thus, result in longer fatigue life. Based upon previous research, no significant differences

would be expected in EHD film thickness [12] or rolling-element

fatigue [25] for lubricants A and B since the only difference between

these fluids is the presence of the additive package in lubricant B.

Test Procedure

The lubricant film thickness was measured using the electrical

capacitance method as described in [3,6]. A schematic is shown in

figure 2. The electrical capacitance between the inner and outer

races of the test bearing is a function of the thickness of the lub-

ricant film and the size of the Hertzian areas of contact (table III).

The capacitance across the test bearing was measured with a capaci-

tance bridge, and the film thickness was read from a calibration curve.

During the tests the capacitance bridge balancing signal was observed

on an ocilloscope. Any short circuiting of the test bearing capacitance

could be observed. This short circuiting was an indication of asperity

contact in the EHD zone. Further details of the measurement techniques

used may be found in [20,26].

Test temperatures were 300, 338, and 393 K (80°, 150°, and 250° F).

Thrust loads applied to the bearing were 90, 445, and 778 N (20, 100,

and 175 lb). The shaft speed was adjusted at speeds from 400 to 15 000

rpm. In all cases the oil flow was approximately the same at 2 to 3

drops per minute, delivered by a mist.

The general test procedure was to begin at top speed and low load

for a given temperature of operation. Then the loads were increased

stepwise to the heaviest load, and held long enough to take a film

thickness reading with the capacitance bridge. The test continued,

repeating the load sequence for each successive lower speed. During

the loading sequence the shaft speed was held constant with an auto-

matic controller. The temperature was controlled to within 3 K (5° F).

For the measurements with lubricant A the test bearing had approxi-

mately 20 to 30 hours of running time on it, which were accumulated dur-

ing checkout and calibration of the test rig. For each of the remaining

test lubricant sequences a new test bearing was used. The tests with

the new bearings were carefully monitored to prevent continued operation

in the asperity contact regime. The test sequence was stopped when it

became clear that the lubricant film was breaking down. This occurred

at the lower speeds and heavy loads.

RESULTS AND DISCUSSION

Effect of Load and Speed

The effect of speed and load on the elastohydrodynamic (EHD) film

for the 20-mm bore angular-contact ball bearing using lubricant C is

shown in figure 3. Figure 3 is a plot of the dimensionless film thickness

H as a function of the dimensionless speed parameter U. The bearing

was run at speeds ranging from 400 to 15 000 rpm, thrust loads of 90,

445, and 778 N (20, 100, and 175 lb) and at temperatures of 300, 338,

and 393 K (80°, 150°, and 250° F). Measured film thicknessed ranged

from 0.025 to 0.51 μm (1 to 20 μin.). The results shown in figure 3 for

lubricant C are representative of the other three lubricants.

Generally, the measured film thickness was more weakly dependent

on speed than theory suggests. At lower temperatures where the

lubricant was more viscous, film thickness was observed to be independent

of speed, and film thickness even decreased at the higher speeds.

For the highest temperature and low speeds the film thickness speed dependence was closest to theoretical prediction.

The measured film thickness decreased slightly with load as predicted by EHD theory. However, in the cases where measured film thickness was independent of speed or decreased with the speed this was not true. In these cases, the measured film thickness increased slightly with load. It is shown later that the principal reason for deviation of the measured values from theory is lubricant starvation. Because of the insignificant difference of EHD film thickness with load, the data for the 90 N (20 lb) thrust load will be presented only in subsequent figures for purposes of clarity.

For synthetic paraffinic lubricants A and B, film thickness measurements were identical for the full range of test parameters. This was expected since lubricants A and B differ only in that B has additives whereas A does not. Thinner films were measured for the tetraester (lubricant C). This was expected since it is less viscous than lubricants A and B. Film thicknesses with the traction fluid (lubricant D) were comparable to those for the lubricants A and B. This is to be expected since the pressure-viscosity coefficient effectively raises the viscosity of this fluid in the contact zone.

Comparison to Classical Theories

The data for the 90 N (20 lb) load points are plotted on dimensionless coordinates in figure 4 and compared to some well known theoretical results. The theories chosen for comparison were those of Grubin [1], Dowson-Higginson [5], Archard and Cowking [4], and Hamrock and Dowson [16]. The respective formulae are as follows:

Grubin's formula $H = 1.95 \ G^{0.73} U^{0.73} W^{-0.091}$

Dowson-Higginson $H = 1.6 \ G^{0.6} U^{0.7} W^{-0.13}$

Archard-Cowking $H = 2.04 \left(1 + \dfrac{2R_x}{3Ry}\right)^{-0.71} G^{0.74} U^{0.74} W^{-0.74}$

Hamrock-Dowson $H = 3.63(1 - e^{-0.68 \ k}) G^{0.49} U^{0.68} W^{-0.73}$

The above equations are plotted in figure 4 for three different values of materials parameter G which represent the maximum, minimum, and average values of G for the experimental conditions. In general there was fair agreement between measured and theoretical film thicknesses at dimensionless speeds less than 10^{-10}. The differences between the film thicknesses calculated with the different EHD theories are less than the scatter in experimentally measured data. The significance of this is to show that some of the past refinements in film thickness prediction theory are unobservable in practice. At the higher speeds of U, between 10^{-10} and 10^{-8}, the correlation to classical isothermal, fully flooded inlet theory is not good.

Thermal Correction and Starvation

Thermal considerations in the inlet zone of the EHD zone [15] and kinematic starvation [27] have an effect on film reduction at higher speeds. Those theories are compared to the present data in figure 5.

A new dimensionless parameter designated "contact lubrication flow number" is introduced in this figure. This is the product of two dimensionless parameters, the materials parameter, G, and the speed parameter, U. This parameter is introduced for this figure because the data of [27] are in this form only.

Theoretical results by Cheng [15] for film thickness reduction

due to inlet shear heating are plotted in figure 5 for the synthetic paraffinic oil. Cheng's theory assumed G = 5000 whereas the values of G for the test conditions range from 1980 to 7620. The average value of G was 3900. Figure 5 shows that inlet shear heating may account for part of the deviation between experiment and isothermal theory at the higher speeds. However, the data show the onset of the deviation at a lower speed, indicating that another mechanism of film thickness reduction is active.

A theoretical model for starvation in a rolling-element bearing was formulated by Y. P. Chiu [27]. Figure 5 shows the results from Chiu's analysis and experimental data [27] which are compared with the experimental data of figure 4. Based upon the correlation between the authors' experimental data and analytical predictions using Chiu's starvation theory, the authors conclude that starvation appears to be the primary cause of the deviation of the experimental data of figure 4 from classical EHD theory.

The lack of adequate lubricant in an EHD contact is affected by the mechanism of lubricant distribution. The mechanism of lubricant distribution was assumed to be any transverse lubricant flow on the rolling track [27]. This lubricant replenishment is aided by higher lubricant-air surface tension, increased ball spacing, and reduced viscosity. Other lubrication replenishment mechanisms are also possible, such as splashing, direct jet deposition, or flow from the cage to balls. All these depend on the particular geometric and kinematic conditions of the bearing [28]. The relative effectiveness of the Y. P. Chiu replenishment model compared to other possible replenishment mechanisms is unknown. However, it is probable that oil film replenishment was occurring due to centrifugal or

direct deposition effects. This is suggested by the continued thickness
of oil film at $GU>10^{-6}$.

Comparison to Other Experiments

Data from NASA sponsored experiments at the Battelle Memorial

Institute [12,13] are compared to the Hamrock-Dowson formula and to the

present experimental data in figure 6. In general, the NASA-BMI data

which were taken using the X-ray film thickness measurement technique

showed film thicknesses greater than theory suggests. The data of

figure 4 showed film thicknesses less than theoretical. The total

amount of scatter in the data from both methods of testing was approxi-

mately equal.

In a discussion to [16], Kunz and Weiner presented data for a

steel ball sliding on a sapphire plate where film thickness was mea-

sured using the optical interferometry method for dimensionless speed

parameters U less than 10^{-11}. Figure 7 shows the comparison of their

results to the experimental data of figure 4. Their data straddled

the theoretically predicted film thickness and had about the same scat-

ter as the present data. The similarity in scatter in film thickness

measurements using X-ray, optical, and capacitance techniques suggests

that the general accuracy of these methods is about the same.

Surface Condition

The bearing running tracks were examined using light and scanning

electron microscopy techniques. In general, no evidence of wear or

surface damage could be seen using light microscopy. However, scanning

electron microscopy revealed many small induced surface defects in the

running track such as shown in figure 8 for lubricant B. The original
grinding marks are clearly visible in the figure which would indicate
that the bearing operated under mostly full EHD film conditions. No com-
parable surface defects could be detected on unrun bearing race surfaces.
These defects were probably induced by wear particles which were genera-
ted during bearing operation and which passed through the ball-race con-
tact zone.

There is a suggestion that the density of the induced defects is
a function of the lubricant. Among lubricants B, C, and D, the density
of the surface defects was highest for lubricant D. (The bearing used
for lubricant A was run for a much longer period of time. Hence, no
valid comparison could be made). The film thickness for a given
operating condition was larger for lubricant D than lubricant C and
equal to lubricant B. This would further suggest that surface defect
density is not a function of EHD film thickness.

SUMMARY AND CONCLUSIONS

Film thicknesses were measured by the capacitance technique for a
thrust loaded 20-mm bore ball bearing. The bearing was mist lubricated
at a rate of 2 to 3 drops per minute. Test parameters were thrust loads
of 90, 445, and 778 N (20, 100, and 175 lb) which gave maximum contact
stresses of 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000
psi) on the inner race, and speeds ranging from 400 to 15 000 rpm.
Tests were conducted with four different lubricants at temperatures of
300, 338, and 393 K (80°, 150°, and 250° F). The lubricants were
(a) synthetic paraffinic, (b) synthetic paraffinic with additives,

(c) a neopentylpolyol ester, and (d) a synthetic cycloaliphatic hydrocarbon traction fluid. The experimental results were compared to theoretical predictions for film thickness. The following results were obtained:

1. For dimensionless speed parameter, U, greater than 10^{-10} there was gross thinning of the lubricant film. Kinematic starvation appears to be the primary cause of reduction in film thickness from classical EHD theory.

2. There was fair agreement between measured and theoretical film thickness at dimensionless speeds less than 10^{-10} using isothermal, fully flooded inlet EHD theory.

3. The differences among the film thicknesses calculated with the different EHD theories are less than the scatter in experimentally measured data. The significance of this is to show that some of the past refinements in film thickness prediction theory are unobservable in practice.

4. The scatter in the experimental data using the capacitance technique was the same as for data taken by other methods. This means that the general accuracy and validity of measurements using X-ray, optical, and capacitance techniques are about the same.

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TABLE I. – TEST BEARING SPECIFICATIONS

Material, balls and races	AISI M-1
Inside diameter, mm (in.)	20 (0.7874)
Outside diameter, mm (in.)	47 (1.8504)
Width, mm (in.).	14 (0.5512)
Pitch diameter, mm (in.)	35.5 (1.4)
Nominal contact angle, deg	17
Inner race curvature, %.	53
Outer race curvature, %.	54
Number of balls.	3
Ball diameter, mm (in.).	7.15 (9/32)
Rockwell C hardness–inner race	62–64
Rockwell C hardness–outer race	62–64
Rockwell C hardness–balls	62–64
Surface finish, rms-races, μm (μin.)	0.15 (6)
Surface finish, rms-balls, μm (μin.)	0.025–0.05 (1–2)
Tolerance specification.	ABEC-5

TABLE II. - LUBRICANT PROPERTIES

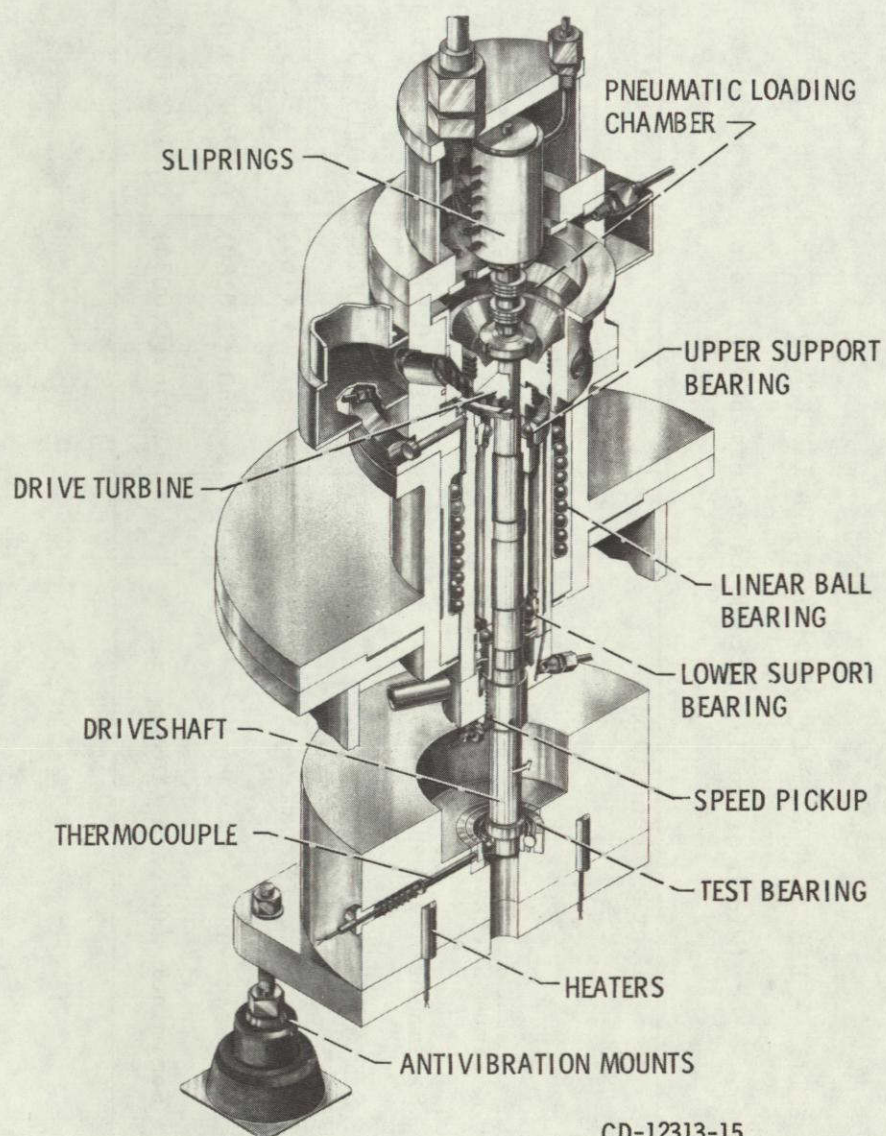| Lubricant	A	B	C	D
Type	Synthetic paraffinic	Synthetic paraffinic	Neopentylpolyol (tetra) ester	Synthetic cycloaliphatic hydrocarbon
Pour point, K	223	223	213	236
Flash point, K	544	544	535	435
Fire point, K	585	585	- -	446
Autogenous ignition, K	677	677	699	599
Density, g/cc				
373 K	0.80	0.80	0.93	0.85
311 K	0.84	0.84	0.98	0.89
Pressure-viscosity exponent, m^2/N				
373 K	1.51×10^{-8}	1.36×10^{-8}	1.00×10^{-8}	1.70×10^{-8}
311 K	1.99×10^{-8}	1.81×10^{-8}	1.37×10^{-8}	3.11×10^{-8}
Kinematic viscosity, cS				
373 K	40	40	5.2	5.6
311 K	447	447	28	34
Surface tension, dyne/cm				
373 K	- -	- -	- -	20
311 K	31	- -	- -	23
Specific heat, J/(kg)(K)	2890 at 473 K	- -	2060 at 373 K	2140 at 373 K
	2650 at 423 K	- -	1913 at 311 K	1860 at 311 K
Thermal conductivity, watt/(m)(K)				
373 K	- -	- -	0.152	0.102
311 K	- -	- -	0.162	0.104
Additives	No	Yes[a]	Yes[b]	Yes[c]

a Organic phosphonate, antiwear additive.
b Oxidation inhibitor, corrosion inhibitor, antiwear additive.
c Antiwear (zinc dialkyldithiophosphate), oxidation inhibitor, antifoam, VI improver (polymethacrylate).

TABLE III. - HERTZIAN CONTACT CONDITIONS AT INNER AND OUTER RACES FOR THREE THRUST LOADS

[Width of rolling track is determined by major axis width.]

Race	Contact condition	Thrust load, N (lbf)		
		90 (20)	445 (100)	778 (175)
Inner	Maximum Hertzian stress, GPa (ksi)	1.28 (185)	2.09 (303)	2.45 (356)
	Semimajor axis, cm (in.)	0.0510 (0.0200)	0.0840 (0.0330)	0.0990 (0.0390)
	Semiminor axis, cm (in.)	0.0066 (0.0026)	0.0110 (0.0043)	0.0130 (0.0051)
Outer	Maximum Hertzian stress, GPa (ksi)	1.13 (164)	1.85 (269)	2.19 (318)
	Semimajor axis, cm (in.)	0.0460 (0.0180)	0.0740 (0.0290)	0.0860 (0.0340)
	Semiminor axis, cm (in.)	0.0086 (0.0034)	0.0140 (0.0055)	0.0170 (0.0065)

PNEUMATIC LOADING CHAMBER

SLIPRINGS

UPPER SUPPORT BEARING

DRIVE TURBINE

LINEAR BALL BEARING

LOWER SUPPORT BEARING

DRIVESHAFT

SPEED PICKUP

THERMOCOUPLE

TEST BEARING

HEATERS

ANTIVIBRATION MOUNTS

CD-12313-15

Figure 1. – EHD film thickness measurement rig.
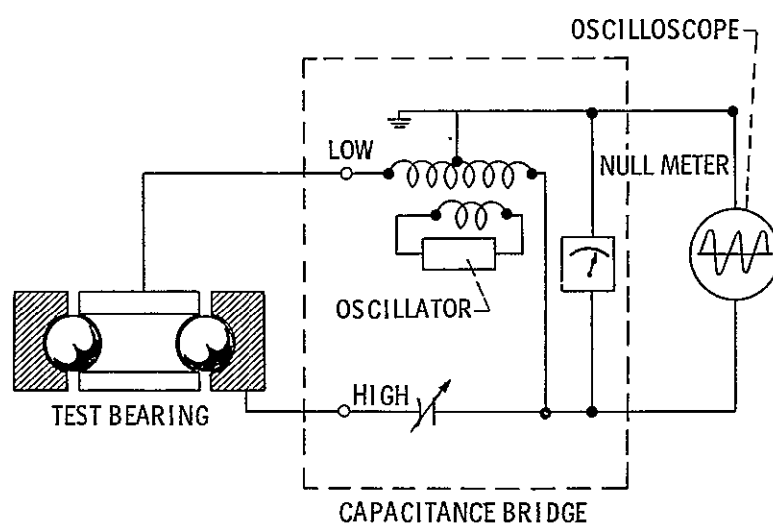
Figure 2. – Film thickness measurement schematic.

DIMENSION-LESS LOAD, W	TEST TEMPER-ATURE, K (°F)	MATERIALS PARAMETER, G
o 3.20×10^{-5} □ 1.37×10^{-4} ◇ 2.24×10^{-4}	300 (80)	3000
◉ 3.20×10^{-5} ◘ 1.37×10^{-4} ◆ 2.24×10^{-4}	338 (150)	2740
● 3.20×10^{-5} ■ 1.37×10^{-4} ◆ 2.24×10^{-4}	393 (250)	1980
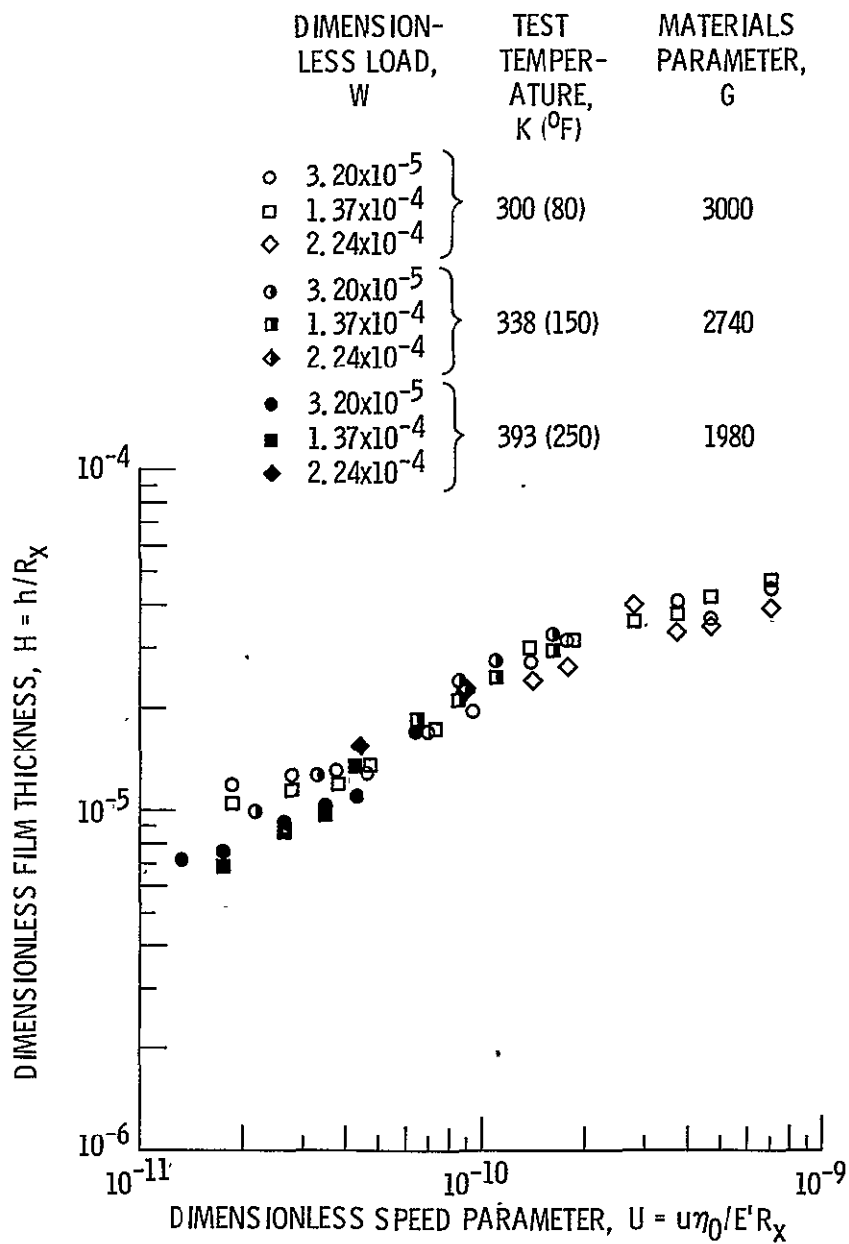
Figure 3. – Effect of speed and load on EHD film thickness. Experimental measurements for 20 mm angular contact bearing. Lubricant C.

	MATERIALS PARAMETER, G	LUBRI- CANT	TEMPER- ATURE, K (°F)
▫	4920	A	
◌	4320	B	300 (80)
◇	3300	C	
○	7620	D	
▫	4030	A	
◌	3660	B	338 (150)
◆	2740	C	
◑	5640	D	
▪	3070	A	
◕	2770	B	393 (250)
◆	1980	C	
●	2770	D	

——— THEORY OF GRUBIN [1]

MATERIALS PARAMETER, G

7620 ———
3900 ———
1980 ———

10^{-3}
10^{-4}
10^{-5}
10^{-6}

(a) THEORY OF GRUBIN [1].

DIMENSIONLESS FILM THICKNESS, $H = h/R_x$

——— THEORY OF DOWSON AND HIGGINSON [5]

10^{-3}
10^{-4}
10^{-5}
10^{-6}

10^{-12} 10^{-11} 10^{-10} 10^{-9} 10^{-8}

DIMENSIONLESS SPEED PARAMETER, $U = u\eta_0/E'R_x$
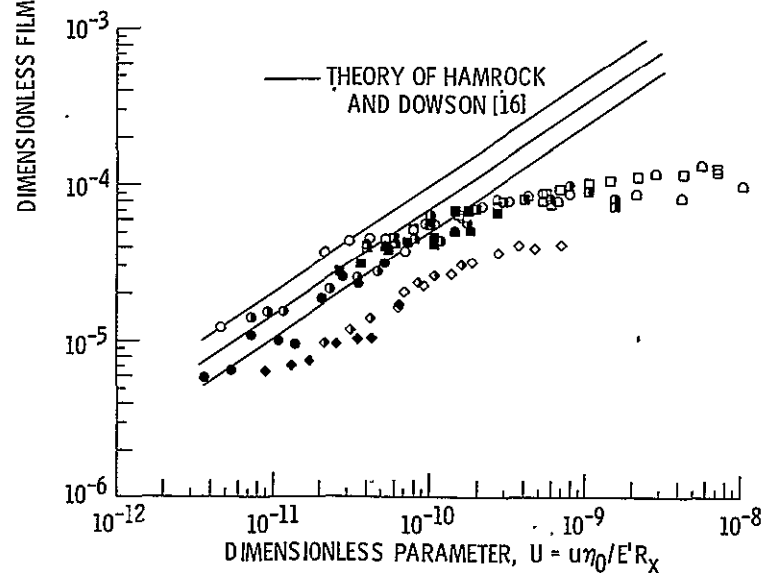
(b) THEORY OF DOWSON AND HIGGINSON [5].
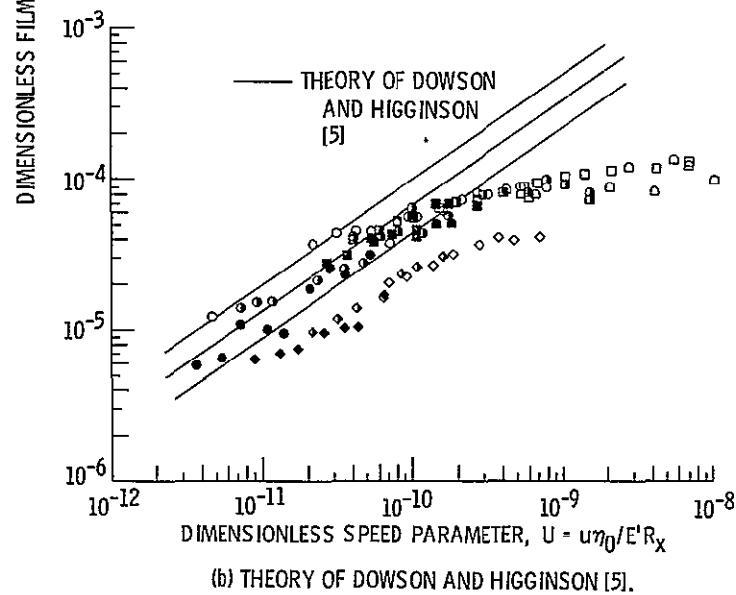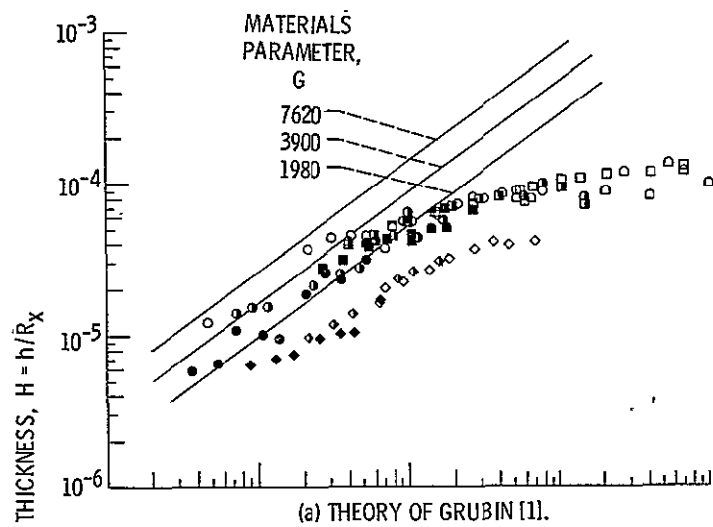
Figure 4. - Comparison of isothermal EHD fully flooded inlet theories to EHD film thickness-measurements for four lubricants. Bearing type, 20-mm bore angular contact; thrust load, 90 N (20 lbf).

	MATERIALS PARAMETER, G	LUBRI- CANT	TEMPER- ATURE, K (°F)
□	4920	A	
◠	4320	B	300 (80)
◇	3300	C	
○	7620	D	
◻	4030	A	
◠	3660	B	338 (150)
◇	2740	C	
○	5640	D	
■	3070	A	
⬟	2770	B	393 (250)
◆	1980	C	
●	2770	D	

——— THEORY OF ARCHARD AND COWKING [4]

MATERIALS
PARAMETER,
G

7620 —
3900 —
1980 —

(c) THEORY OF ARCHARD AND COWKING [4].

——— THEORY OF HAMROCK
AND DOWSON [16]

DIMENSIONLESS FILM THICKNESS, $H = h/R_x$

DIMENSIONLESS PARAMETER, $U = u\eta_0/E'R_x$
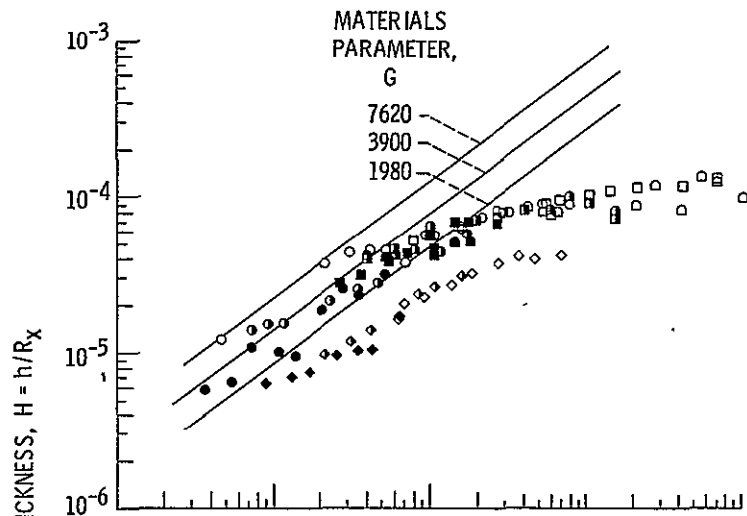
(d) THEORY OF HAMROCK AND DOWSON [16].

Figure 4. - Concluded.
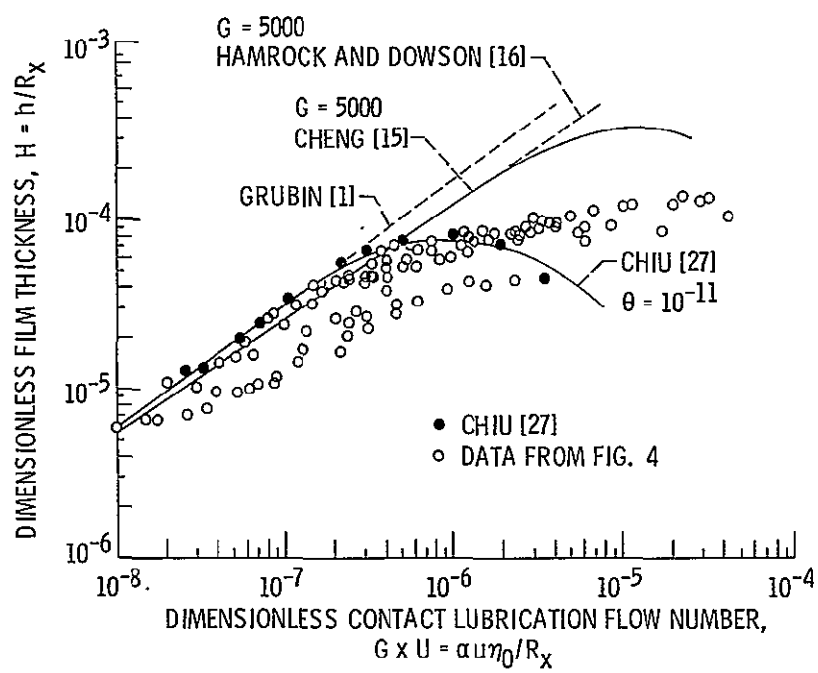
Figure 5. – Theoretical effect of kinematic starvation and inlet shear heating on film thickness and comparison with experiment.

	NASA BMI X-RAY DATA [12 AND 13]		THEORY (HAMROCK AND DOWSON) [16]
	MATERIALS PARAMETER, G	LUBRICANT	
▲	2440	SYNTHETIC PARAFFINIC	— — — —
■	2440	TYPE II ESTER	— — — —
◆	4200	POLYPHENYL ETHER	— — · —
●	5550	FLUOROCARBON	— — — —
○	EXPERIMENTAL DATA FROM FIG. 4		————
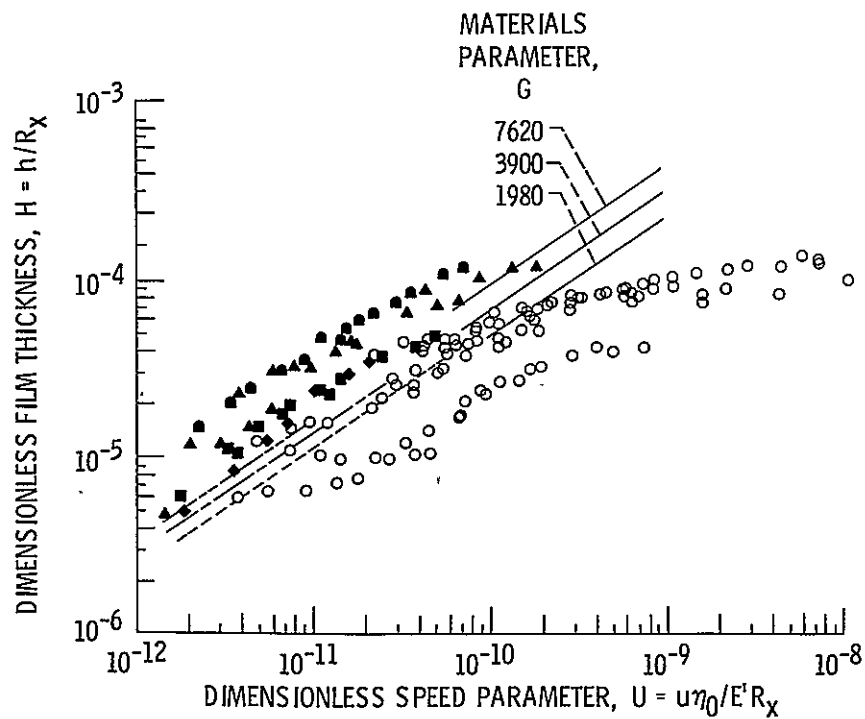
Figure 6. – Comparison with X-ray film thickness measurements.

EXPERIMENT (KUNZ AND WEINER) [16]

LOAD		MATERIALS PARAMETER, G	LUBRICANT	THEORY (HAMROCK AND DOWSON) [16]
1.24×10^{-6}	9.29×10^{-6}			
△	▲	4507	POLYALKYL AROMATIC	– – – – –
◁	◀	8874	SYNTHETIC HYDROCARBON	– · – · –
◻	◼	5107	MODIFIED POLYALKYL ETHER	– ·· – ·· –
○			EXPERIMENTAL DATA FROM FIG. 4	————
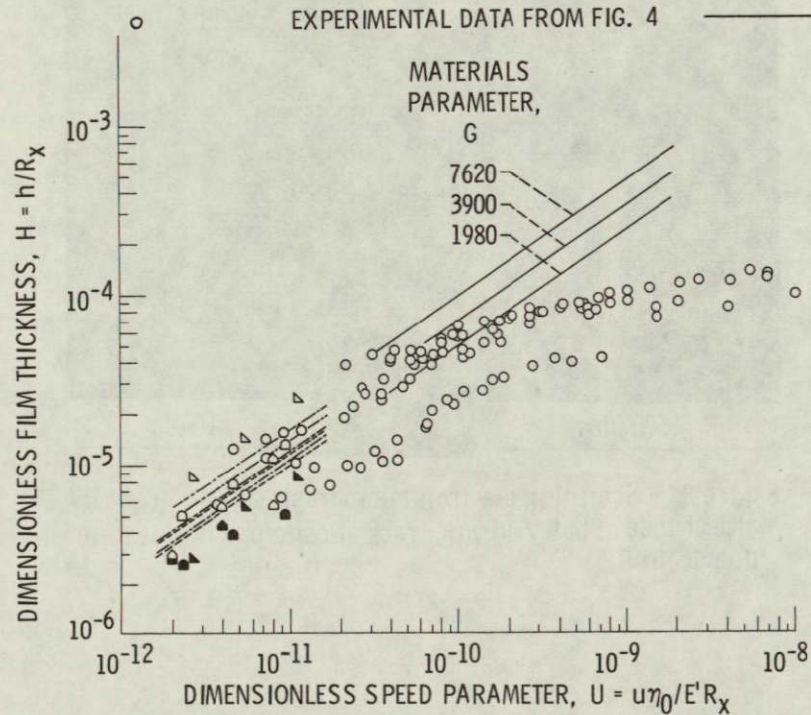
Figure 7. – Comparison with optical film thickness measurements under pure sliding.

500 µin.   10 µm
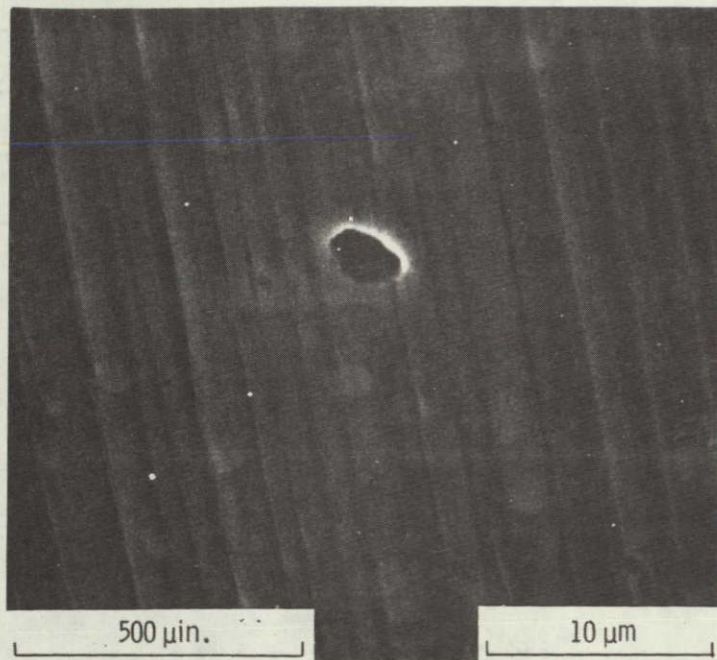
Figure 8. - Scanning electron micrograph showing typical appearance of ball running track at conclusion of testing (lubricant B).

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15 Supplementary Notes

16. Abstract

Elastohydrodynamic film thickness was measured for a 20-mm ball bearing using the capacitance technique. The bearing was thrust loaded to 90, 448, and 778 N (20, 100, and 175 lb). The corresponding maximum stresses on the inner race were 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000 psi). Test speeds ranged from 400 to 14 000 rpm. Film thickness measurements were taken with four different lubricants: (a) synthetic paraffinic, (b) synthetic paraffinic with additives, (c) neopentylpolyol (tetra) ester meeting MIL-L-23699A specifications, and (d) synthetic cycloaliphatic hydrocarbon traction fluid. The test bearing was mist lubricated. Test temperatures were 300, 338, and 393 K. The measured results were compared to theoretical predictions using the formulae of Grubin, Archard and Cowking, Dowson and Higginson, and Hamrock and Dowson. There was good agreement with theory at low dimensionless speed, but the film was much smaller than theory predicts at higher speeds. This was due to kinematic starvation and inlet shear heating effects. Comparisons with Chiu's theory on starvation and Cheng's theory on inlet shear heating were made.

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